

Article

Comparative Analysis of Direct Drop-In Fluid Replacement for a Centrifugal Compression System

Jordan Dickenson, James R. Bull, Jovana Radulovic *  and James M. Buick 

School of Electrical and Mechanical Engineering, University of Portsmouth, Portsmouth PO1 3DJ, UK; jordan.dickenson@port.ac.uk (J.D.); james.bull@port.ac.uk (J.R.B.); james.buick@port.ac.uk (J.M.B.)

* Correspondence: jovana.radulovic@port.ac.uk

Abstract

The phasing out of high-GWP refrigerants and the growing diversity of working fluids used across heat pumps, refrigeration systems, and closed-cycle power applications have made drop-in fluid replacement a question of significant practical interest. Centrifugal compressors are designed around the thermophysical properties of a specific fluid, and the performance penalty is incurred when working fluid is replaced without redesigning the impeller. This study presents a CFD comparison of direct drop-in fluid replacement in a fixed geometry centrifugal compression system. Eight working fluids that span the property range relevant to current drop-in substitutions are evaluated: air, nitrogen, argon, carbon dioxide, R22, R134a, R1234yf, and R1234ze(E). A reference centrifugal impeller was reconstructed in ANSYS BladeGen, meshed in ANSYS TurboGrid using the Automatic Topology and Meshing method, and simulated in ANSYS CFX (2024 R2) as a single periodic passage with Frozen Rotor interfaces and Spalart–Allmaras turbulence closure. Performance maps were generated for each fluid across a range of rotational speeds and mass flow rates, with a common inlet reference condition applied across all cases to isolate the influence of fluid properties from an inlet state. The resulting dataset enables a like-for-like comparison of pressure ratio, efficiency, and shaft power requirement, providing a basis for assessing the aerodynamic implications of drop-in fluid substitution in centrifugal compression systems. Air, nitrogen, argon and carbon dioxide achieved similar peak efficiencies (~88%) and comparable pressure ratios (PR), indicating they can be used as drop-in substitutes without performance loss. Refrigerants R1234yf and R1234ze(E) matched R134a in efficiency (peak ~90%) while offering higher pressure ratios and significantly lower power requirements at peak efficiency. At 20,000 RPM and a mass flow rate of 2 kg/s, compared to a PR of 1.45 for air, nitrogen, carbon dioxide and argon achieved PRs of 1.4, 1.9 and 2.4, respectively. At the same settings, R134a and R1234 refrigerants reached PRs of 5 and 6, respectively. The power requirement was $\sim 8 \times 10^4$ W for air and similar fluids, and $\sim 11 \times 10^4$ W for refrigerants.



Academic Editor: Li Xi

Received: 12 June 2026

Revised: 13 July 2026

Accepted: 14 July 2026

Published: 27 July 2026

Copyright: © 2026 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the [Creative Commons Attribution \(CC BY\)](https://creativecommons.org/licenses/by/4.0/) license.

Keywords: drop-in fluid replacements; compressor; CFD simulation; efficiency; refrigerants

1. Introduction

Thermophysical properties of working fluids govern the aerodynamic, thermodynamic, and mechanical behaviour of turbomachinery. Classical theory shows that fluid properties determine velocity triangles, work transfer, compressibility effects, and loss mechanisms, forming the basis for key performance parameters such as specific speed, Mach number, Reynolds number, and polytropic efficiency [1,2]. In centrifugal compressors,

these properties are central to stage design, directly influencing pressure ratio, efficiency, operating range, and stability [3,4].

The importance of working fluid properties has driven an increasing interest in drop-in fluid replacement. Drop-in replacements can operate in existing equipment with minimal modification to enable use of environmentally friendly or higher-performance fluids. This significantly reduces cost and downtime compared to full system redesigns and also allows compliance with environmental regulations [5,6].

Many researchers have investigated the performance of working fluid replacements for phased-out fluids such as R134a, both experimentally and theoretically [7–9]. Hmood et al. provided a systematic literature review of drop-in replacements, finding that R1234yf was the most suitable alternative to R134a [10]. Belman et al. experimentally evaluated the performance of R1234ze, R513a and R516a as replacements for R134a and identified R1234ze as the most suitable candidate [11]. Similarly, Yatağanbaba et al. conducted a thermodynamic assessment comparing several low-GWP refrigerants and reported that R1234yf and R1234ze demonstrated promising performance with small system modifications [12]. Navarro-Esbrí et al. performed both theoretical and experimental analyses on R1234yf, indicating comparable cooling capacities to R134a but with slightly reduced efficiency under certain operating conditions [13]. Mota-Babiloni et al. further investigated HFO-based refrigerants and concluded that blends such as R513a offer a balanced compromise between energy efficiency and environmental impact [14]. Jawad et al. performed a thermodynamic analysis of several low-GWP refrigerants in a vapour compression refrigeration cycle under varying condenser and evaporator pressures, comparing their performance against R134a [15]. The study found that R152a exhibited the highest coefficient of performance (COP) and that although R600a and R1234yf provided slightly lower COP values, their extremely low GWP makes them environmentally suitable candidates for specific applications. Kizilboga et al. conducted a comprehensive review of low-GWP refrigerants, evaluating performance, safety, and environmental impact, and identified R-152a, R-290, and R-600a as the most promising replacements for R134a due to their thermodynamic performance, despite flammability concerns [16]. The study also showed that although R-1234yf and R-1234ze(E) exhibit lower flammability and significantly reduced GWP, their performance is generally lower than R134a, limiting their effectiveness as direct substitutes.

These studies evaluated the performance of replacement fluids through an entire system, often with full-cycle thermodynamic analysis and system optimisation. However, comparatively fewer studies isolate the compressor itself, despite it being the primary component governing thermodynamic work input and efficiency. Tian et al. investigated the influence of air, argon, CO₂, CO and helium on compressor cascade aerodynamics using numerical analysis [17]. They found that variations in thermophysical properties, particularly specific heat ratio, density, and compressibility, significantly alter cascade performance by affecting shock strength, flow separation, and loss generation. It was noted that differences in density influenced mass flow behaviour and blade loading, consistent with established turbomachinery theory [18]. Meroni et al. also investigated both variation in fluid performance and impeller optimisation using air, R134a and CO₂ [19]. It was demonstrated that fluid choice significantly alters optimal stage loading, impeller sizing, and achievable pressure ratios, highlighting the strong coupling between fluid properties and compressor design. Wang and Yan investigated the performance variations of a two-stage centrifugal compressor when transitioning from R134a to the low-GWP alternatives R1234ze(E) and R1234yf using a combined CFD and 1D meanline modelling approach [20]. The results indicated that lower inlet speed of sound leads to higher corrected speeds, increasing pressure ratios for R1234yf and R1234ze compared to R134a, while variations in density and latent heat influence mass flow rate. Overall, the study emphasises that work-

ing fluid substitution significantly impacts compressor aerodynamic and thermodynamic behaviour, requiring careful consideration for drop-in systems.

In summary, the literature suggests a clear need for more targeted investigation into compressor performance under drop-in conditions. While prior studies demonstrate that working fluid properties strongly influence aerodynamic behaviour and that system performance can be maintained with suitable substitutes, there remains limited understanding of how these fluids comparatively perform within a fixed compressor geometry. As such, the present study addresses this gap by evaluating a range of working fluids within a constant compressor geometry including both refrigerants (R22, R134a, R1234ze, and R1234yf) and non-refrigerants (nitrogen, argon, air, and CO₂). The conventional refrigerants R22 and R134a provide established performance benchmarks, given their extensive use in vapour compression systems, and R1234yf and R1234ze are selected as recommended low-GWP drop-in replacements for R134a. Air and nitrogen are widely used reference fluids in compressor studies due to their well-characterised ideal gas behaviour and practical relevance. Argon, with a higher molecular weight and lower specific heat ratio, allows assessment of compressibility and density effects under more inert, monatomic gas conditions. Finally, CO₂ combines relatively simple gas behaviour with high density and notable real gas effects, making it representative of fluids used in both industrial gas compression and advanced refrigeration or power cycle applications. By keeping the geometry fixed and varying only the working fluid and compressor operational conditions, this work aims to isolate the direct impact of fluid properties on compressor performance, providing insight into the feasibility and limitations of true drop-in substitution in centrifugal compressor applications.

2. Methodology

2.1. Geometry

The compressor geometry was created in ANSYS 2024 R2 BladeGen, reconstructed from a technical drawing [21]. As the original source did not provide raw coordinate data, the blade profile coordinates were extracted from the drawing using a digital point-extraction tool. The digitised coordinates were then imported into BladeGen to reconstruct the meridional flow-path profile, including both the main blades and the splitter blades. The key geometry parameters are summarised in Table 1, and the reconstructed meridional profile is shown in Figure 1. The 16 meridional point coordinates used to define the flow-path boundary in BladeGen are listed in Table 2; points 4 and 15 represent the leading edge of the splitter blade only and are not used on the main blade meridional plane. The splitter blades were reconstructed using the same digitised coordinate approach as the main blades; no lean or sweep was applied, and the blade angle and thickness distributions were kept identical to those of the main blades. The final rotor configuration included 13 main blades and 13 splitter blades, consistent with the reference geometry [21].

Table 1. Key compressor geometry parameters.

Parameter	Value
Blade type	Centrifugal impeller
Number of main blades	13
Number of splitter blades	13
Blade reconstruction method	Digitised coordinate import
Blade lean/sweep	None applied

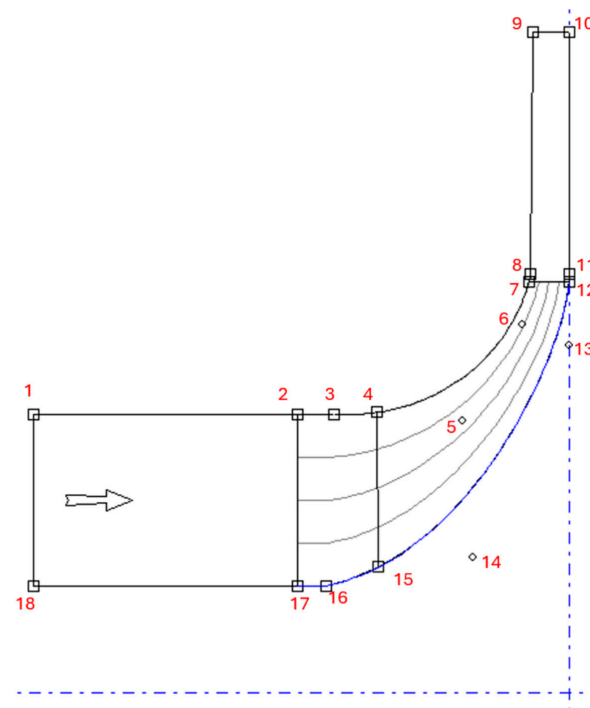


Figure 1. Compressor meridional profile reconstructed in ANSYS BladeGen. Point labels correspond to the coordinates listed in Table 2.

Table 2. Meridional point coordinates corresponding to Figure 1.

Point	X (mm)	Y (mm)
1	−150	78.0
2	−76.0	78.0
3	−66.0	78.0
4 (Splitter Blade Only)	−53.4	35.2
5	−30.0	76.0
6	−13.0	103
7	−11.4	115
8	−11.0	117
9	−10.2	185
10	0.00	185
11	0.0	117
12	0.738	115
13	0.00	97.0
14	−27.0	38.0
15 (Splitter Blade Only)	−53.8	78.5
16	−68.0	30.0
17	−76.0	30.0
18	−150	30.0

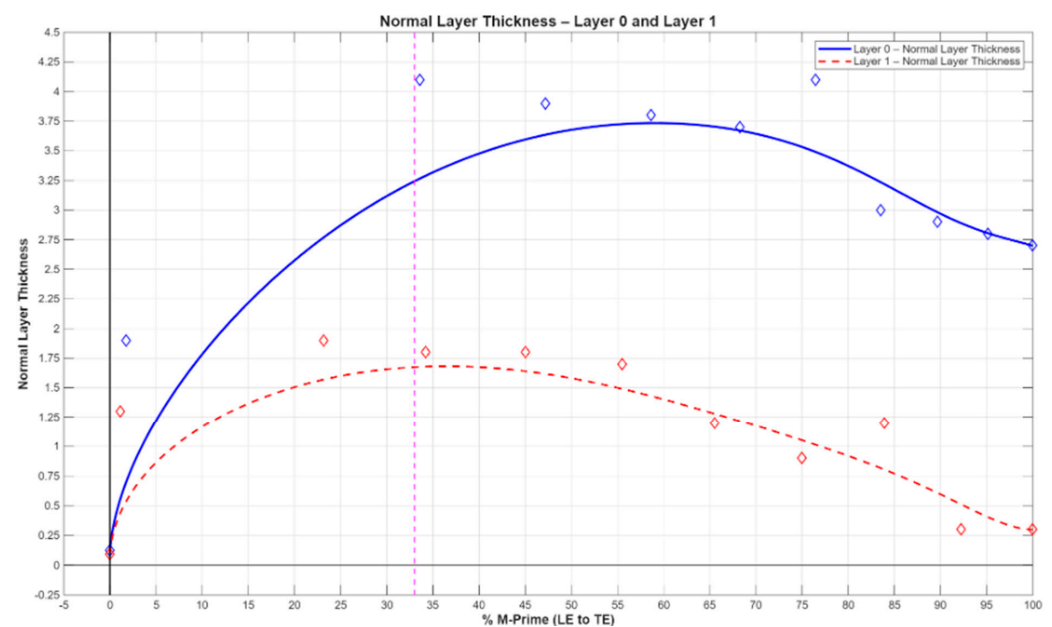
The compressor features two distinct blade layers: Layer 0 corresponds to the blades at the hub and Layer 1 to the blade at the shroud. The blade angle and thickness distributions for each layer were prescribed independently along the hub-to-shroud meridional direction, as given in Tables 3 and 4 and illustrated in Figures 2 and 3 respectively. Lean and sweep were not applied; for the splitter blade, the thickness and blade angle remained unchanged.

Table 3. Layer 1 and Layer 0 angle distribution of the main blade along normalised meridional arc length (M').

Layer 1		Layer 0	
Percentage Along Blade Edge	Angle ($^{\circ}$)	Percentage Along Blade Edge	Angle ($^{\circ}$)
0	−37	0	−63.0
62.7	−37	15.7	−63.0
63.7	−2.5	42.3	−56.0
83.6	−2.5	73.7	−41.0
90.9	−37	82.8	−38.0
100	−37	100	−38.0

Table 4. Layer 1 and Layer 0 thickness distribution of main blade along meridional arc length (M).

Layer 1		Layer 0	
Percentage Along Blade Edge	Thickness (mm)	M (mm)	Thickness (mm)
0	0.01	0	0.12
1.12	1.30	1.76	1.90
23.2	1.90	33.6	4.10
34.2	1.80	47.2	3.90
45.0	1.80	58.7	3.80
55.5	1.70	68.3	3.70
65.5	1.20	76.5	4.10
75.0	0.90	83.5	3.00
83.9	1.20	89.7	2.90
92.2	0.30	95.1	2.80
100	0.30	100	2.70

**Figure 2.** Layer 0 and 1 normal layer thickness against the percentage length of black from leading edge to trailing edge, with splitter blade LE being denoted by the pink line.

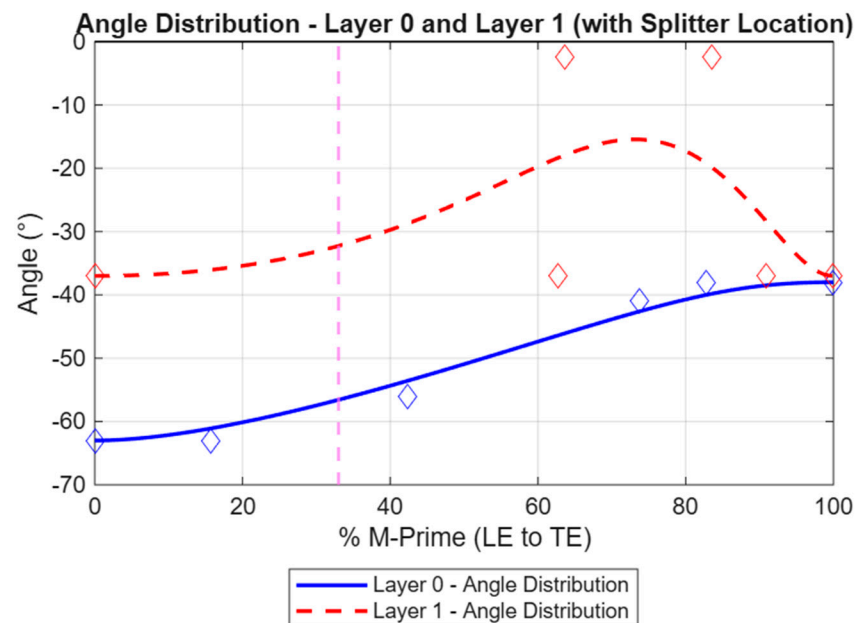


Figure 3. Layer 0 and 1 blade angle against the percentage length of black from leading edge to trailing edge, with splitter blade LE being denoted by the pink line.

2.2. Solver Setup

The simulation was constructed in ANSYS CFX using a three-domain configuration: Inlet, Passage Main, and Outlet. Each domain was defined as a separate fluid region and connected via general connection interfaces at the domain boundaries. The interaction between the rotating impeller passage and the stationary inlet and outlet domains was managed using the Frozen Rotor interface model, which is appropriate for steady-state stage analyses where the relative position of rotor and stator is fixed. The Frozen Rotor model preserves circumferential non-uniformities at the interface and avoids the averaging inherent in the Stage (mixing-plane) approach, making it better suited to single-passage periodic simulations.

Rotational periodicity was imposed at the circumferential boundaries of the passage to represent the full annular geometry at a fraction of the computational cost. A single passage was therefore simulated for each operating point, with periodic boundary conditions replicating the effect of the remaining 12 passages.

2.3. Mesh Generation

The computational mesh was generated in ANSYS TurboGrid using the Automatic Topology and Meshing (ATM) method. ATM was selected in preference to an unstructured tetrahedral approach, such as the Delaunay-based edge/face/volume meshing used in the reference study [21], because it produces a predominantly hexahedral, body-fitted mesh aligned with the blade passage geometry. This structured topology is better suited to capturing the thin boundary layers and strong streamline curvature present in centrifugal compressor passages, and reduces numerical diffusion compared with tetrahedral elements.

A global element size factor of 1.0 was applied. Boundary-layer refinement was introduced using a mesh-proportional method with a factor base of 3, which progressively clusters elements towards the blade, hub, and shroud surfaces to resolve near-wall gradients adequately. The mesh had approximately 180,000 elements in total. The mesh generation settings are summarised in Table 5, and the resulting blade-to-blade mesh topology is shown in Figure 4. The domain configuration is illustrated in Figure 5, and the solver settings are listed in Table 6. A mesh independence study was conducted to confirm that

the selected mesh density produced results insensitive to further refinement; the results are presented in Section 2.6.

Table 5. CFX solver and domain configuration.

Setting	Value
Solver	ANSYS CFX
Simulation type	Steady state
Domain configuration	3 domains (Inlet/Main/Outlet)
Rotor–stator interface	Frozen Rotor
Circumferential periodicity	Rotational (1 passage)
Timescale factor	1.0

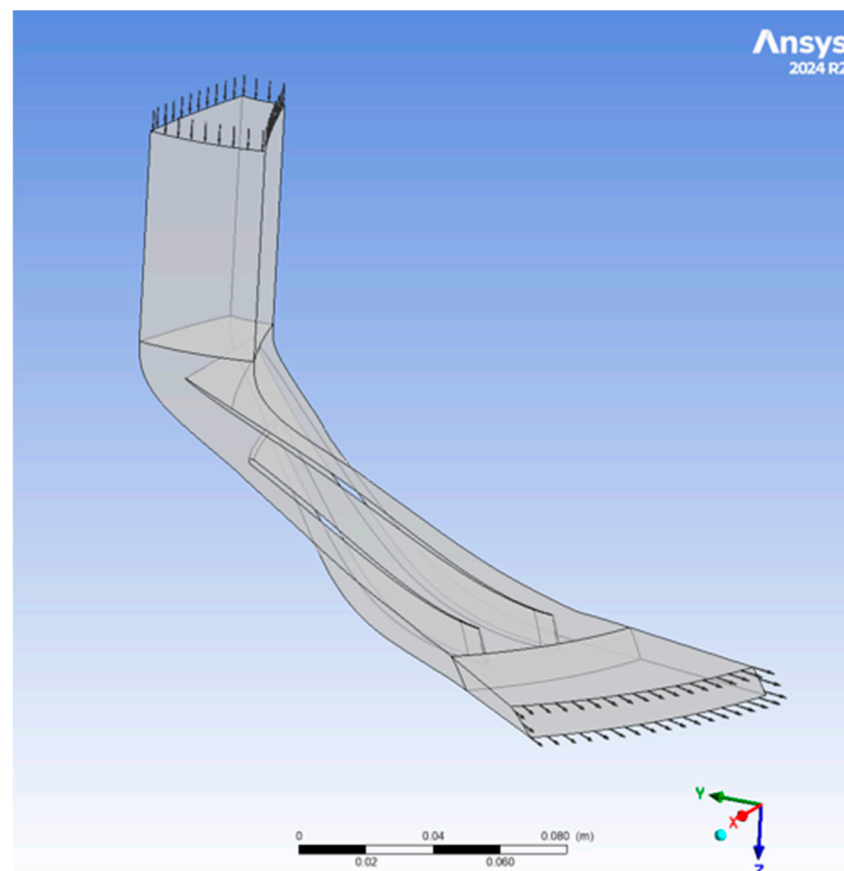


Figure 4. ANSYS CFX three-domain configuration showing the Inlet, Passage Main, and Outlet regions with the Frozen Rotor interface locations shown.

Table 6. Mesh generation settings.

Setting	Value
Meshing method	ATM (TurboGrid)
Global size factor	1.0
BL refinement method	Mesh-proportional
BL factor base	3
Total element count	≈180,000

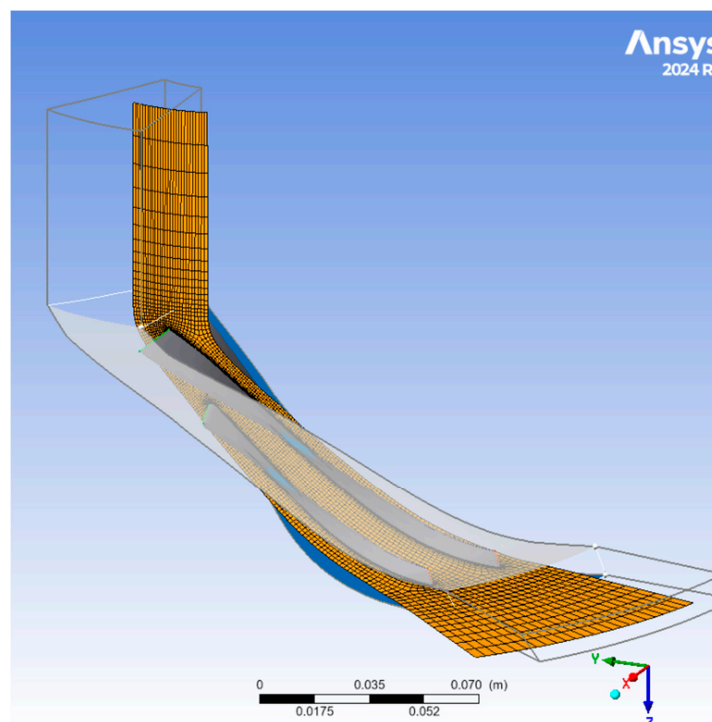


Figure 5. Blade view of the computational mesh generated in ANSYS TurboGrid, showing the structured hexahedral elements within the impeller passage.

2.4. Physical Models and Boundary Conditions

2.4.1. Fluid Properties and Thermodynamic Modelling

This study investigates the performance of the compressor geometry across eight working fluids: air, nitrogen (N_2), argon (Ar), R1234yf, R1234ze(E), R22, R134a, and carbon dioxide (CO_2). These fluids span a broad range of molecular weights, thermodynamic properties, and application contexts, enabling a systematic comparison of compressor performance under different fluid conditions using a single fixed geometry. The eight working fluids investigated in this study were selected as discussed in Section 1, encompassing both refrigerants (R22, R134a, R1234yf, and R1234ze(E)) and non-refrigerants (air, nitrogen, argon, and CO_2) to cover a broad range of thermophysical parameters and application contexts.

For air, nitrogen, argon, and CO_2 fluid properties were represented using the ideal gas equation of state. These fluids behave approximately as ideal gases at the pressure ratios and temperatures met in this study. For the refrigerants R1234yf, R1234ze(E), R134a, and R22, real gas behaviour departs significantly from the ideal gas assumption, particularly at elevated pressures and near the saturation boundary. These four fluids were therefore modelled using Real Gas Property (RGP) table files.

RGP files were generated using the ANSYS CFX Real Gas Property generator, with thermodynamic data sourced from the NIST REFPROP database. The tabulated properties cover the full expected range of temperatures and pressures within the compressor domain for each fluid, including density, specific heat capacity, dynamic viscosity, thermal conductivity, and speed of sound. Using pre-tabulated real gas properties avoids the computational overhead of in situ equation-of-state evaluation at each solver iteration while maintaining thermodynamic accuracy. Each RGP table was generated with a sufficient margin above and below the expected operating conditions to prevent out-of-range interpolation errors during simulation.

2.4.2. Turbulence Modelling and Wall Treatment

The Spalart–Allmaras (SA) one-equation turbulence model was selected for all simulations. The SA model solves a single transport equation for the modified eddy viscosity $\tilde{\nu}$, making it computationally efficient relative to two-equation models while retaining adequate accuracy for wall-bounded, attached boundary-layer flows. The model was originally developed for aerodynamic applications and performs well in flows where adverse pressure gradients remain mild and boundary-layer separation is limited, which is consistent with the on-design operating conditions investigated in this study.

Wall treatment was applied using the standard near-wall formulation in ANSYS CFX. The mesh was designed to place the first cell centroid within the viscous sublayer, consistent with the near-wall resolution requirements of the SA turbulence model. Details of the resulting y^+ distribution and verification of the near-wall mesh quality are presented in Section 2.6.

2.4.3. Boundary and Domain Conditions

A total pressure inlet boundary condition was applied at the inlet domain face, with turbulence intensity set to 3% and the flow direction specified as normal to the boundary. The reference pressure for all domains was set to 1 Atmosphere, with a relative inlet total pressure of 1 Atmosphere. At the outlet boundary, a mass flow rate condition was prescribed. The inlet total temperature was set to 300 K for all fluids as a common reference condition, enabling a controlled comparison of compressor performance attributable to fluid properties alone rather than differences in inlet thermodynamic state.

All solid surfaces were treated as no-slip, adiabatic walls. Rotational speed was applied to the impeller domain, with the hub and blade surfaces co-rotating and the shroud treated as a counter-rotating wall at zero absolute velocity to approximate the stationary casing. The boundary conditions are summarised in Table 7.

Table 7. Summary of boundary conditions, applied identically for all fluids unless otherwise noted.

Boundary	Type	Value/Setting
Inlet	Total pressure	1 Atmosphere (relative)
Inlet turbulence intensity	Percentage	3%
Inlet flow direction	Normal to boundary	-
Inlet total temperature	Fixed value	300 K
Outlet	Mass flow rate	1–3.5 kg/s
Blade/hub surfaces	No-slip wall	Adiabatic
Shroud	Counter-rotating wall	Zero absolute velocity
Boundary-to-boundary interface and rotational model	Frozen Rotor	-

2.5. Operating Conditions

Simulations were performed across a matrix of rotational speeds and mass flow rates to characterise the compressor performance map for each working fluid. Rotational speeds from 10,000 RPM to 20,000 RPM were evaluated in 2000 RPM increments, giving six speed lines per fluid. At each speed, the mass flow rate at the outlet boundary was generally varied between 1 kg/s and 3.5 kg/s; however, results were omitted where the compressor was in choke and the results for the non-refrigerants were extended down to 0.5 kg/s to provide a fuller description of the compressor performance.

The operating range was selected to be consistent with the bounds used in the reference study [21], enabling direct comparison of pressure ratio and isentropic efficiency as a function of mass flow rate at each speed line. The same speed and mass flow rate values were applied across all eight fluids; however, differences in fluid density, speed of

sound, and molecular weight result in different aerodynamic loading and compressor map characteristics for each fluid relative to the air baseline. Table 8 reflects the properties of each fluid at the inlet for the selected operating conditions.

Table 8. Operating condition matrix showing the thermophysical properties of the fluids at the inlet; properties taken from REFPROP Nist 10 [22].

Working Fluid	Density (kg/m ³)	Compressibility	Specific Heat Ratio	Molecular Weight (g/mol)	GWP
Argon	3.25	0.99	1.67	39.9	0
Nitrogen	2.28	1.00	1.40	28.0	0
Carbon dioxide	3.61	0.99	1.29	44.0	1
Air	2.36	0.99	1.40	28.9	0
R1234yf	9.69	0.96	1.10	114	0
R1234ze(E)	9.73	0.95	1.10	114	6
R134a	8.64	0.96	1.12	102	1430
R22	7.24	0.97	1.18	86.5	1810

Simulations were run to a steady-state RMS residual target of 1×10^{-6} for all equations, with mass conservation imbalances maintained within 1×10^{-5} across all domains, consistent with the convergence criteria adopted in Dickenson et al. [23]. Additional checks for solution stability were performed by monitoring inlet and outlet boundary quantities and the polytropic and isentropic efficiency values over the final iterations, with a case accepted only once these had stabilised alongside the residual and conservation targets.

2.6. Mesh Independence Study

A mesh independence study was performed using CO₂ as the working fluid to ensure that the simulation results were insensitive to the spatial discretisation. Eight mesh densities were evaluated by varying the global element size factor in TurboGrid from 0.5 to 3.0. For each mesh, the pressure ratio, corrected mass flow rate, polytropic efficiency, isentropic efficiency, and passage-exit Mach number were recorded at a fixed operating point. The results are presented in Table 8.

The solution shows clear convergence with increasing size factor (finer mesh). Between the size factor 1.0 mesh ($\approx 170,000$ elements, selected mesh, denoted) and the next finer mesh at size factor 0.75, the pressure ratio changes by less than 0.08% and the isentropic efficiency by less than 0.90% (Table 9). Coarser meshes at size factors of 1.75 and above show deviations exceeding 1% in efficiency compared to the selected mesh. In addition to global performance convergence, near-wall resolution was assessed to ensure consistency with the turbulence modelling assumptions. For the selected size factor 1.0 mesh, wall y^+ values remained below 1 across the blade surfaces, satisfying the near-wall resolution requirements of the Spalart–Allmaras turbulence model. A representative wall y^+ contour plot for CO₂ showed a consistently low distribution throughout the passage, while the corresponding histogram confirmed that the majority of wall-adjacent cells had y^+ values below 1 (Figure 6). On this basis, the size factor 1.0 mesh was judged to provide a satisfactory balance between solution accuracy and computational cost and was adopted for all later simulations.

Model validation was carried out by comparing simulation results against the SRV2-O performance data reported in Dickenson et al. [23], in which the same geometry and ANSYS CFX framework were employed, building on the original experimental dataset for the SRV2-O impeller reported by Eisenlohr et al. [24]. At 40,000 RPM, efficiency predictions agreed with the experimental data to within $\pm 2.5\%$, providing confidence in the suitability

of the selected mesh and solver configuration for the present study. The shaft power input was computed in ANSYS CFX as

$$P = -\tau_{z,blade} * N_{blade} * \omega$$

where $\tau_{z,blade}$ is the torque about the axis of rotation evaluated on the blade surface of the single simulated passage, $N_{blade} = 13$ is the number of main blades (scaling the single passage result to the full impeller), and ω is the rotational speed. The negative sign accounts for the sign convention in which the solver reports the reaction torque exerted by the fluid on the blade, opposite in sign to the shaft work delivered to the fluid.

Table 9. Mesh independence study results. Percentage differences (Δ) are computed compared to the selected mesh at size factor 1.0.

	Element Count	PR	Corrected m (kg/s)	η Polytropic	η Isentropic	Δ PR (%)	$\Delta\eta$ Isentropic (%)
0.75	76,840	1.2540	0.187	0.7819	0.7782	0.17%	0.39%
1.00	167,440	1.2550	0.187	0.7888	0.7852	0.25%	0.51%
1.25	334,455	1.2542	0.187	0.7879	0.7843	0.18%	0.39%
1.50	595,280	1.2545	0.187	0.7889	0.7853	0.21%	0.52%
1.75	913,445	1.2517	0.187	0.7808	0.7771	0.01%	0.53%
2.00	131,726	1.2516	0.187	0.7803	0.7765	0.02%	0.60%
3.00	4,400,640	1.2519	0.187	0.7849	0.7812	0.00%	0.00%

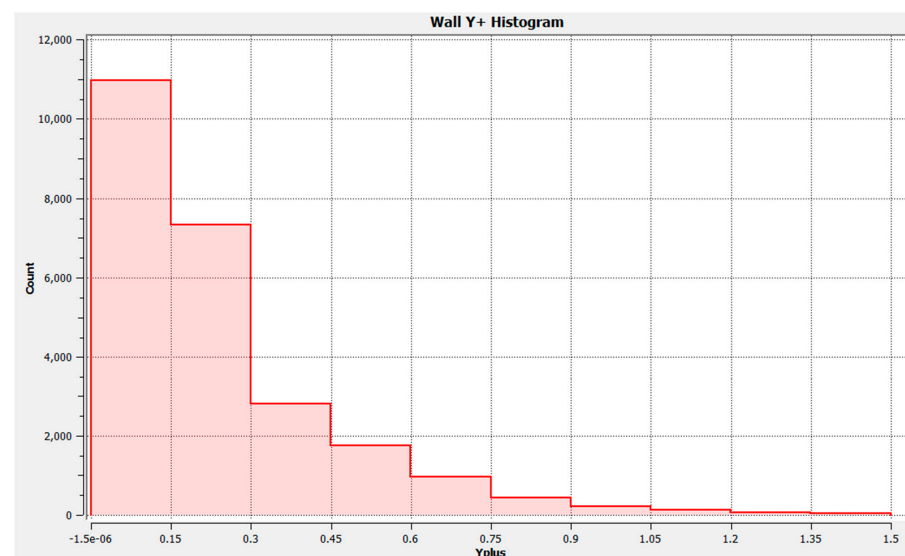


Figure 6. Histogram of wall y^+ distribution of carbon dioxide simulation after convergence, at 10,000 RPM and 1.5 kg/s mass flow rate.

Figure 7 compares the simulated polytropic and isentropic efficiency curves at 40,000 RPM against the experimental total-to-total efficiency reported for the SRV2-O geometry [23]. The simulation reproduces both the location of the efficiency peak (≈ 2.1 kg/s) and its magnitude (polytropic ≈ 0.87 , isentropic ≈ 0.84) to within $\pm 2.5\%$ of the experimental value, confirming that the mesh, solver settings, and turbulence closure adopted here reproduce the correct performance trend as well as the correct peak value for this geometry.

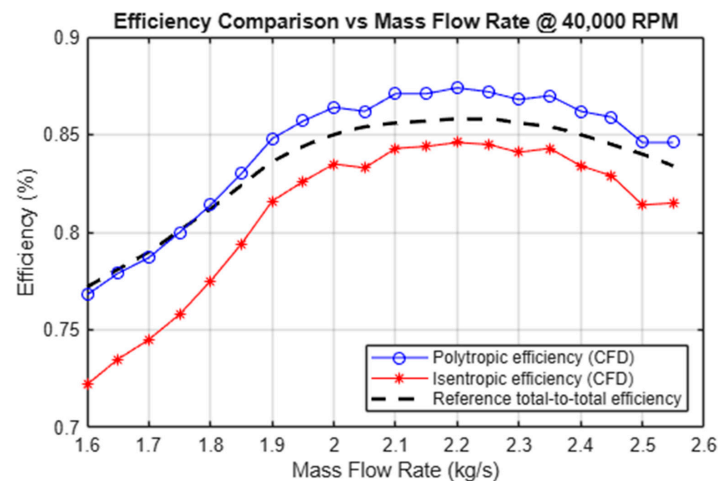


Figure 7. Comparison of isentropic efficiency and polytropic efficiency against mass flow rate from Dickenson et al. [23], with experimental total-to-total efficiency against mass flow rate collected from Eisenlohr et al. [24].

While the validated operating point corresponds to the design speed of the SRV2-O geometry, the mesh independence and near-wall (y^+) verification described above confirm that the same mesh and turbulence model configuration remain valid at the reduced rotational speeds considered in this study.

3. Results

Before examining the global performance maps, flow field contours were taken from CFX Post to confirm that the simulations produce physically consistent results. Figure 8 shows static pressure and relative velocity distributions within the blade passage for air and R1234yf at a common operating condition of 1 kg/s and 10,000 RPM. In both cases, the pressure increases smoothly from the leading edge to the trailing edge of the impeller passage, and the relative velocity field shows a consistent deceleration through the passage consistent with the expected behaviour of a centrifugal compressor operating within its stable range. The higher density of R1234yf relative to air results in a more pronounced pressure gradient across the passage at the same mass flow rate and rotational speed, consistent with the thermophysical property differences summarised in Table 8. No regions of significant flow separation or unphysical behaviour were identified, providing confidence in the integrity of the CFD results presented in the following sections.

Figures 9–11 show the variations in the pressure ratio, isentropic efficiency and power requirements across the range of mass flow rates, for the different rotational speeds considered. For each of the fluids, the pressure ratio, shown in Figure 9, increases with the rotational speed, as would be expected, with the rate of increase also increasing with the rotational speed. The argon, nitrogen, CO₂ and air show the pressure ratio decreasing with the mass flow rate, indicating that the compressor is approaching choke at the higher mass flow rates and the peak pressure ratio is achieved around, or below, the lowest mass flow rate considered. By contrast, the refrigerants all show little variation in the pressure ratio over the range of mass flow rates, indicating that the compressor is operating around its peak value and that the peak is broader than observed for the other fluids. Additionally, there is also a difference between the pressure ratios obtained between the two groups of fluids. At the highest speed, 20,000 RPM, the pressure ratio for the refrigerants is higher than the other fluids, with R1234yf and R1234ze(E) producing a pressure ratio of approximately 6. Conversely, the results suggest that the other fluids achieve a lower pressure ratio, even allowing for the peak value being outside the mass flow range considered, with

argon, for example, achieving a maximum pressure ratio of 2.5. At 10,000 RPM, a similar trend is observed, although here the differences are smaller due to the lower pressure ratios produced at these speeds, with all pressure ratios being less than 2.

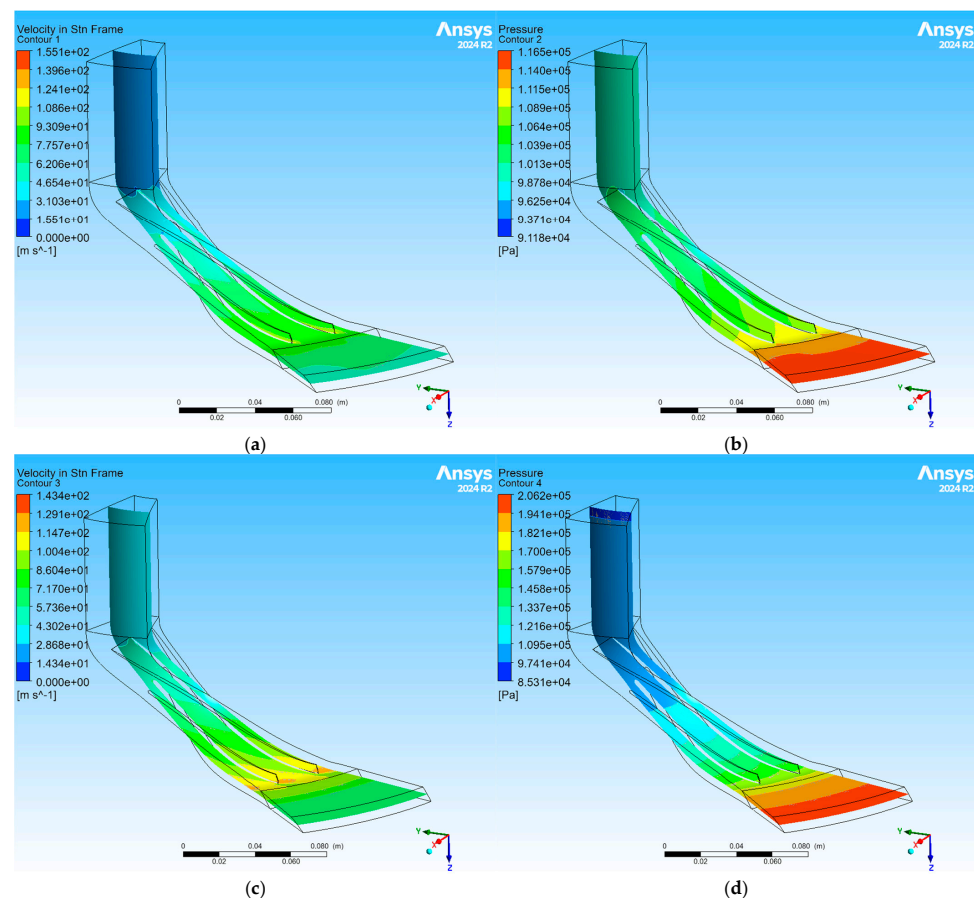


Figure 8. (a) Velocity contour plot of air at 1 kg/s and 10,000 RPM, (b) pressure contour plot of air at 1 kg/s and 10,000 RPM, (c) velocity contour plot of R1234YF at 1 kg/s and 10,000 RPM, (d) pressure contour plot of R1234YF at 1 kg/s and 10,000 RPM.

Figure 10 presents the results for the isentropic efficiency. For each of the fluids considered, the maximum value of the isentropic efficiency is approximately the same at just under 0.9. The results for argon, nitrogen, CO₂ and air show the position of the peak isentropic efficiency increasing with mass flow rate and also show a rapid drop in the isentropic efficiency at the highest mass flow rates, consistent with the observation from Figure 8 that for these gases the compressor is operating close to stall, particularly for the lower mass flow rates. In contrast, the results for the refrigerants with the peak isentropic efficiency occur close to, or above, the range of mass flow rates considered. For the refrigerants, at any mass flow rate, the isentropic efficiency is seen to be increasing with a decreasing rotational speed. This is consistent with the peak in the isentropic efficiency increasing with mass flow rate, as was observed for the other cases, and the maximum efficiency being approximately constant over all of the fluids considered.

Figure 11 shows the variation in the power requirements. In most cases the power requirements increase with both the mass flow rate and the rotational speed of the compressor. The exceptions are for argon, nitrogen, CO₂ and air, where the power requirement levels off or starts to decrease at the higher mass flow rates, where the isentropic efficiency indicates the compressor is approaching stall, as seen in Figure 10, and the pressure ratio drops close to 1, as seen in Figure 9.

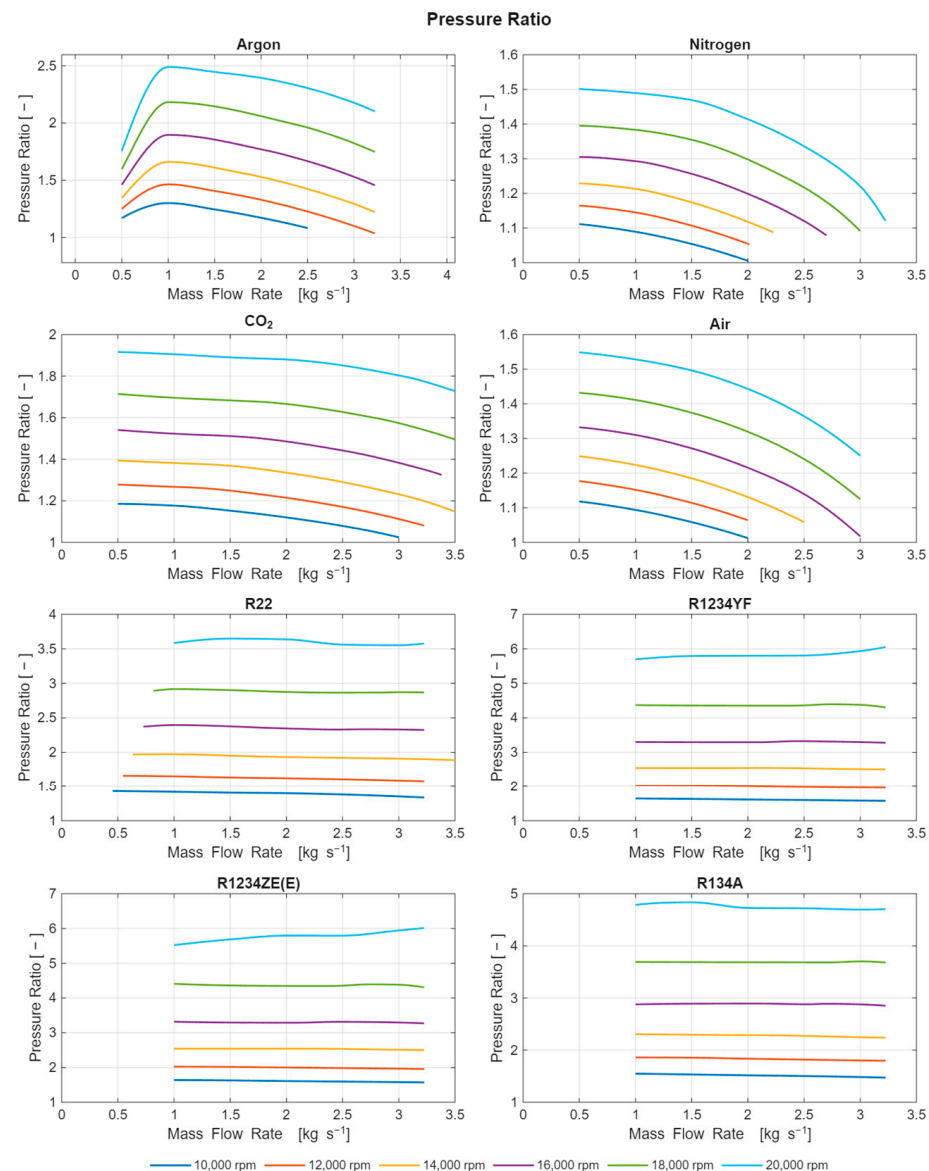


Figure 9. Pressure ratio variation with mass flow rate at rotation speeds of 10,000, 12,000, 14,000, 16,000, 18,000 and 20,000 RPM for argon, nitrogen, CO₂, air, R22, R1234yf, R1234ze(E) and R134a.

Due to similarities in their thermophysical properties, trends identified in performance curves for air, nitrogen, CO₂ and argon are comparable. For the same RPM and mass flow settings, the pressure ratio achieved by air is virtually the same as that of nitrogen (1.25–1.52 for air and 1.22–1.49 for mass flow rates of 3–1 kg/s at 20,000 RPM), and lower than that of CO₂ (1.80–1.91) and argon (2.18–2.49). However, the peak efficiency of all low-density fluids is practically the same, with a difference of less than 0.0045 in the maximum value recorded over the full mass flow rate range at 20,000 RPM, although there are some differences in the settings (mass flow rate for the given RPM) required for the max isentropic efficiency. Power curves also show consistent trend across all low-density fluids: similarly lower power requirement for air (73.1 kW for 2 kg/s at 20,000 RPM) and nitrogen (73.1 kW), and higher for argon (83.2 kW) and CO₂ (90.8).

For the same mass flow rate settings, with the peak occurring at a higher RPM, refrigerant fluids achieve higher pressure ratios, in particular R1234yf and R1234ze(E), which achieved pressure ratios in the range of 5.70–5.93 and 5.52–5.94 respectively for a mass flow rate in the range of 3–1 kg/s at 20,000 RPM. Unlike low-density fluids, isentropic efficiency increases with increasing mass flow rate, indicating that improved performance

is achieved at higher mass flow rates, without adverse effect on the overall pressure ratio achieved. However, low-density fluids require less compression work; hence, higher power requirement is necessitated for refrigerant fluid compression. As seen in Figure 9 the position of the pressure peaks changes with the speed and mass flow rate, and so the extra power requirements also change with the power requirements for R1234yf and R1234ze(E), exceeding a factor of 2 for some settings, compared to the non-refrigerants.

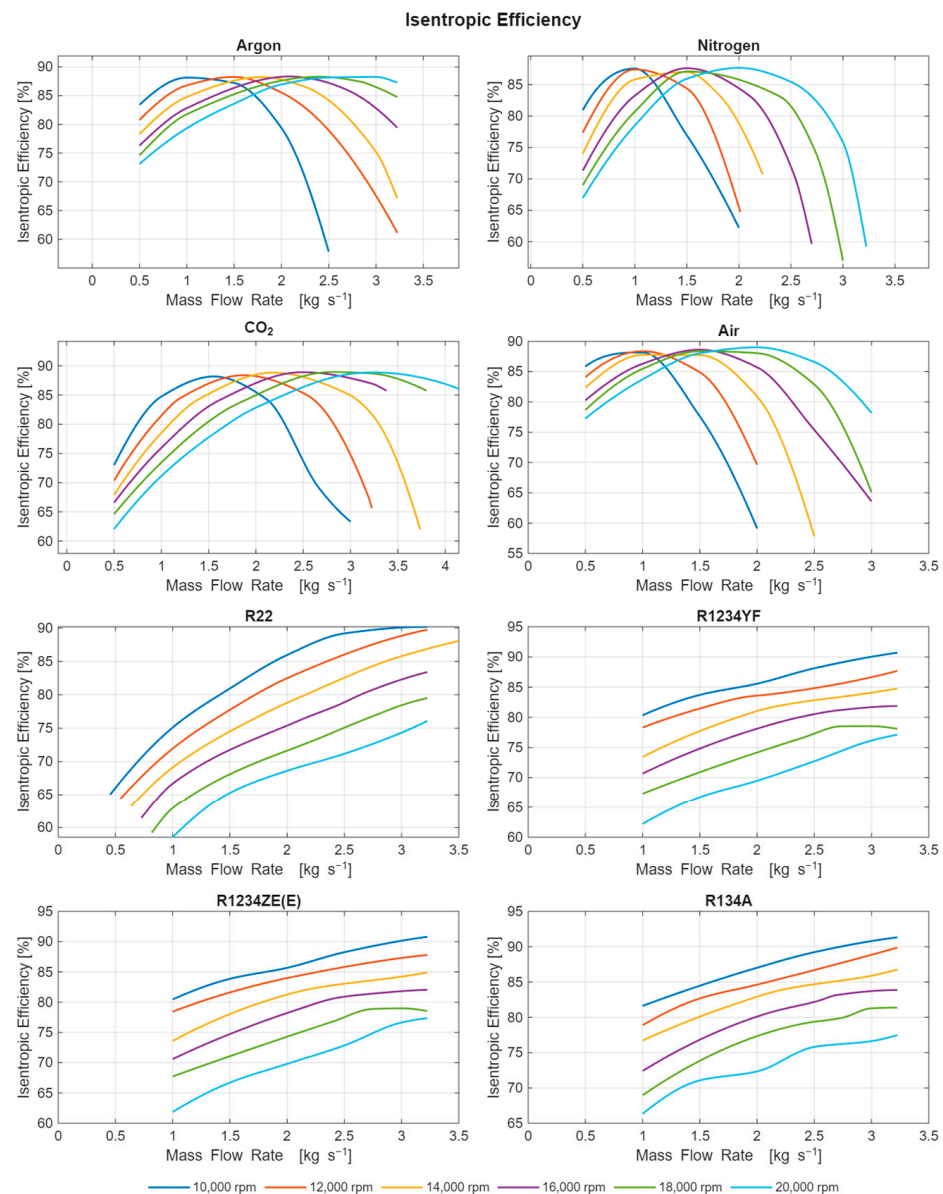


Figure 10. Variation in isentropic efficiency against mass flow rate at rotation speeds of 10,000, 12,000, 14,000, 16,000, 18,000 and 20,000 RPM for argon, nitrogen, CO₂, air, R22, R1234yf, R1234ze(E) and R134a.

Overall, the data indicate that refrigerants can be effectively used as drop-in replacement fluids in studies of the air-compressor system, achieving higher pressure ratios with equivalent efficiency and a relatively minor increase in the power requirement, as evidenced in Figures 12–14.

Performance envelopes collectively demonstrate that at a given speed, refrigerants exhibit optimal performance at higher mass flow rates at or beyond the maximum considered here, compared to air and nitrogen, reflecting their higher density, which requires greater mass throughput to achieve the aerodynamic loading the impeller was designed for. In

contrast, air and nitrogen reach peak performance at lower mass flow and are dependent on the rotational speed and fall within the operating range considered, but demonstrate broader operating ranges due to their higher speed of sound delaying the onset of choke. CO₂ displays intermediate behaviour, combining a moderately elevated optimal mass flow rate with increased sensitivity to off-design operation due to its real gas compressibility effects. An increase in rotational speed raises the pressure ratio for all fluids (Figure 12) but does not alter the relative positioning of the curves, confirming that the observed trends are primarily governed by fluid properties rather than rotational speed. An equivalent conclusion can be drawn for Figures 13 and 14; at a given rotational speed, lighter working fluids reach optimal performance at lower mass flow rates, whereas denser gases require higher flow rates and exhibit broader operating regions. This confirms that fluid density and thermophysical properties govern both the location and sharpness of optimal operating points, with refrigerants exhibiting the most compact yet sensitive operating regions.

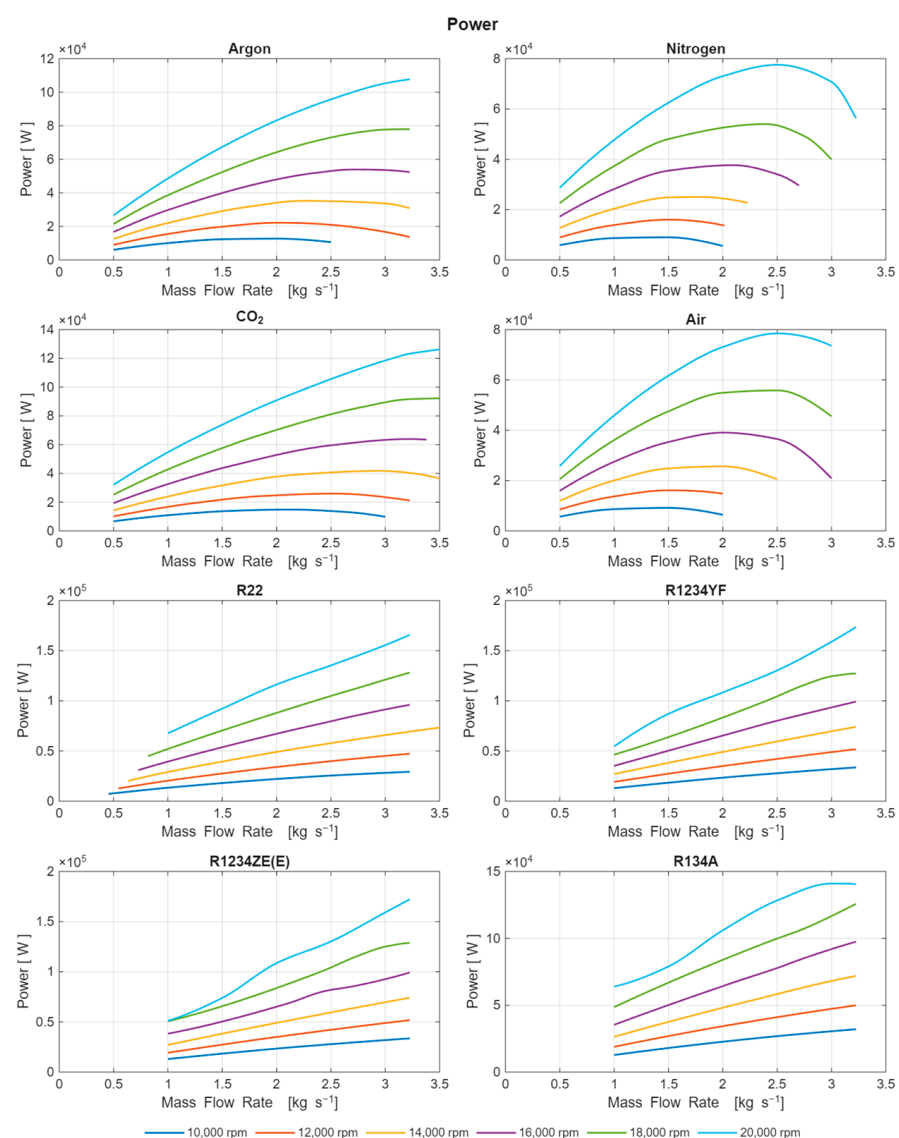


Figure 11. Power requirement variation with mass flow rate at rotation speeds of 10,000, 12,000, 14,000, 16,000, 18,000 and 20,000 RPM for argon, nitrogen, CO₂, air, R22, R1234yf, R1234ze(E) and R134a.

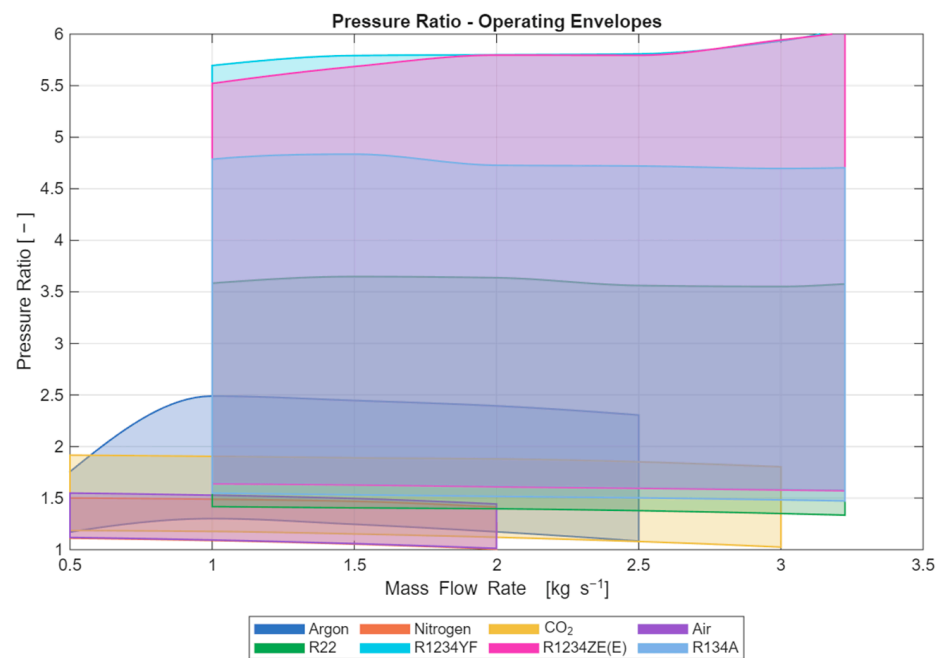


Figure 12. Operating performance envelopes showing pressure ratio against mass flow rate for all eight working fluids. Shaded regions represent the operating envelope bounded by the 10,000 and 20,000 RPM speed lines.

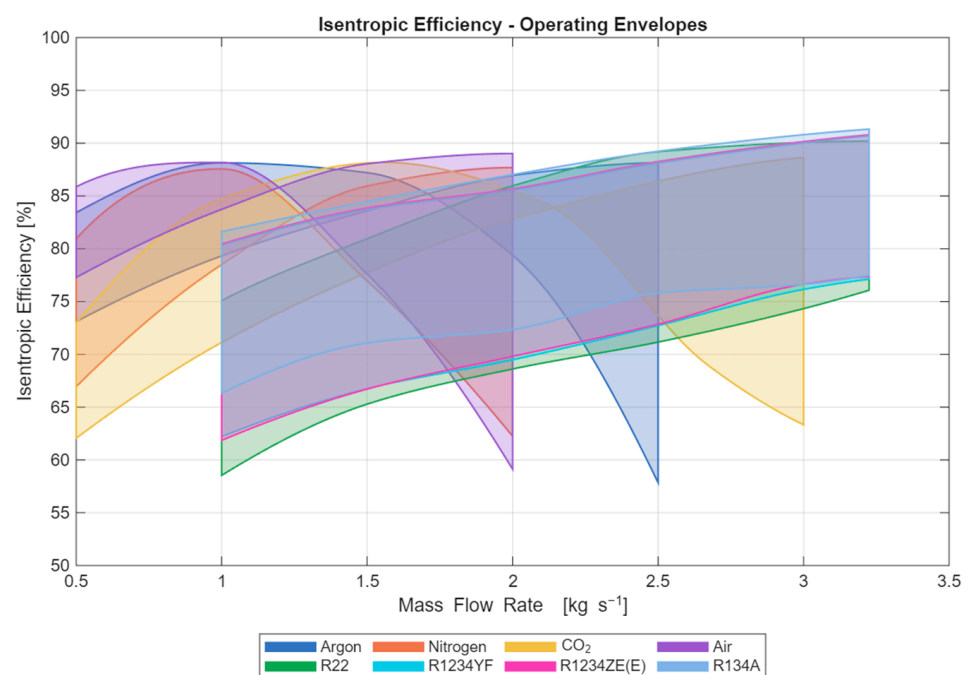


Figure 13. Operating performance envelopes showing isentropic efficiency against mass flow rate for all eight working fluids. Shaded regions represent the operating envelope bounded by the 10,000 and 20,000 RPM speed lines.

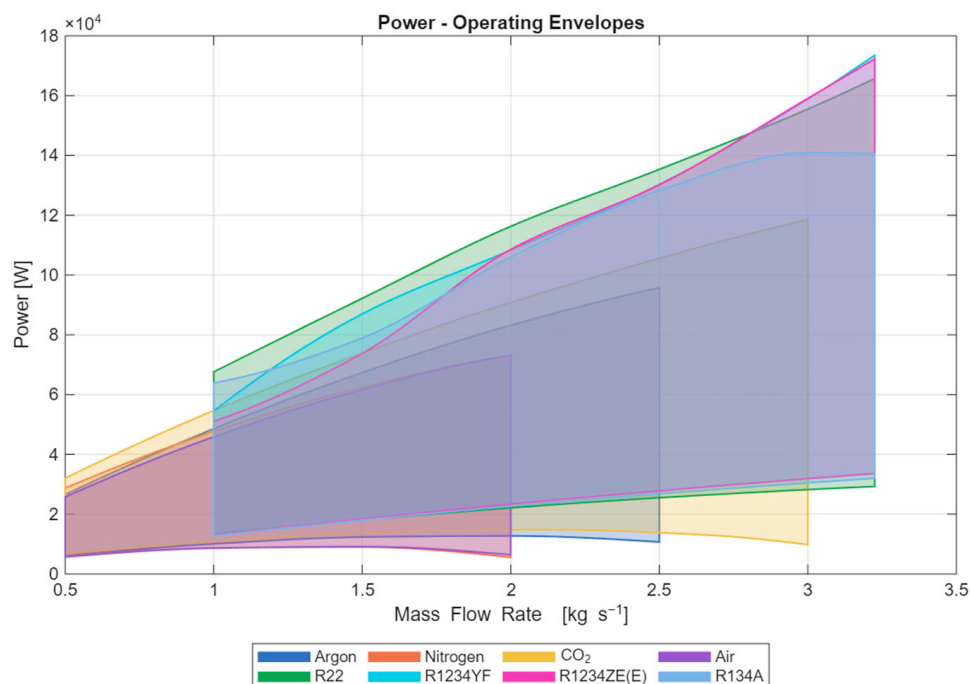


Figure 14. Operating performance envelopes showing shaft power requirement against mass flow rate for all eight working fluids. Shaded regions represent the operating envelope bounded by the 10,000 and 20,000 RPM speed lines.

4. Discussion

The differences in compressor performance across the eight working fluids investigated can be attributed to differences in thermophysical properties; molecular weight, density, and specific heat ratio. The impeller geometry was originally designed around air, optimised for a specific volumetric flow rate and velocity triangle distribution at the impeller inlet and outlet. When a denser fluid is substituted, the same prescribed mass flow rate at the outlet boundary corresponds to a lower volumetric flow rate through the passage, altering the velocity triangles and shifting the aerodynamic loading onto the impeller. Therefore, the peak in pressure ratio and isentropic efficiency shifts to higher mass flow rates for denser fluids, as greater mass throughput is required to achieve the aerodynamic loading the impeller was designed for. This is consistent with the findings of Meroni et al. [16], who demonstrated that fluid choice significantly alters optimal stage loading within a fixed compressor geometry. The prescribed mass flow outlet boundary condition ensures a consistent and controlled comparison across all fluids, isolating the direct influence of thermophysical properties on compressor performance within the same fixed geometry. While each fluid operates at a different volumetric flow rate for the same prescribed mass flow, this reflects the real-world condition of direct drop-in substitution, where the mass flow demands of the system remain fixed regardless of fluid choice.

The pressure ratio achieved at a fixed operating point correlates almost perfectly with molecular weight across all eight fluids tested, as shown in Figure 15. At 20,000 RPM and a mass flow rate of 3.225 kg/s, nitrogen (molecular weight 28.0 g/mol) achieves a pressure ratio of 1.25, rising through argon (39.9 g/mol, PR 2.10) and CO₂ (44.0 g/mol, PR 2.55), to R22 (86.5 g/mol, PR 6.15) and R1234yf and R1234ze(E) (114.0 g/mol, PR 11.1). The higher-pressure ratios achieved by heavier fluids are from their lower specific heat ratio, which allows a given work input to produce a greater pressure rise. This supports the findings of Wang and Yan [17], who identified that the lower speed of sound in R1234yf and R1234ze(E) leads to higher corrected speeds and pressure ratios relative to R134a.

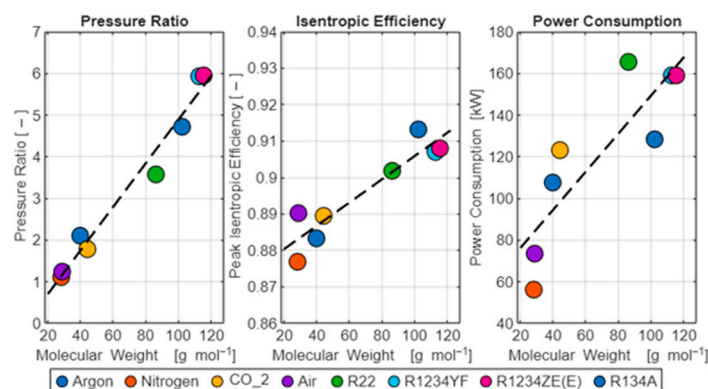


Figure 15. Correlation of pressure ratio at 20,000 RPM and 3.225 kg/s, peak isentropic efficiency within the whole range, and power consumption at 20,000 RPM and 3.225 kg/s against the respective molecular weight of each fluid.

Isentropic efficiency follows the same molecular weight trend, although with a less strong correlation. When each fluid is assessed at its own optimal operating point, peak isentropic efficiencies converge into two broad groups. The non-refrigerant fluids all achieve peak efficiencies of approximately 0.88 across their respective optimal speed lines, while the refrigerants R1234yf, R1234ze(E), and R134a consistently achieve higher peak efficiencies of approximately 0.92 to 0.93. Within the two groups there is not a clear proportionality between the isentropic efficiency and the molecular weight. The relatively narrow spread of approximately 5% across fluids spanning a molecular weight range of 28 to 114 g/mol suggests that aerodynamic loss mechanisms within the fixed impeller geometry are relatively insensitive to fluid thermophysical properties when each fluid operates at its own optimal conditions, consistent with Meroni et al. [16]. Notably, the refrigerants achieve this higher peak efficiency at significantly higher mass flow rates, with R1234yf and R1234ze(E) peaking outside of the studied range, compared to approximately 2.5 to 3.0 kg/s for the non-refrigerant group.

Power consumption is seen to increase closely with the molecular weight of the fluids, with the exception of R22, consistent with the pressure ratio trend. R1234yf and R1234ze(E) require the greatest shaft power, followed by CO₂, then argon, with air and nitrogen requiring the least, directly reflecting their respective densities and the greater thermodynamic work input required to compress heavier fluids. R22 exhibits the highest power consumption of all fluids tested; however, as it is included purely as a comparison baseline it is not considered further in this analysis. The power gap between fluid groups widens with increasing rotational speed, consistent with higher blade tip speeds amplifying the influence of fluid density differences on velocity triangles. At 20,000 RPM and 3.225 kg/s, R1234yf and R1234ze(E) require shaft power of approximately 172 kW and 173 kW respectively, compared to 108 kW for argon and 123 kW for CO₂, representing a power increase of approximately 60% over the non-refrigerant group at this operating point.

With respect to drop-in feasibility, the results demonstrate two distinct outcomes depending on the fluid group considered. For the non-refrigerant fluids, air, nitrogen, argon, and CO₂ all achieve peak isentropic efficiencies of approximately 0.88 within the mass flow range investigated, with pressure ratios broadly consistent with the air baseline for which the impeller was designed. This suggests that non-refrigerant fluids with similar properties to air are compatible with the fixed geometry without significant performance penalty, making them viable substitutes in applications where refrigerant properties are not required. CO₂ represents a notable case within this group, achieving marginally higher pressure ratios than air and nitrogen due to its higher molecular weight and real gas

compressibility effects, while maintaining comparable peak efficiency, making it a strong candidate for industrial gas compression applications within this geometry.

For the refrigerant group, R1234yf and R1234ze(E) demonstrate comparable peak isentropic efficiency to R134a within the fixed geometry investigated, confirming that aerodynamic compatibility is maintained under direct fluid substitution without geometric modification. Both fluids achieve higher pressure ratios than R134a at equivalent rotational speeds, suggesting that drop-in substitution with these low-GWP alternatives would not result in a performance penalty. The trade-off between pressure ratio performance and power demand is more favourable when fluids are compared at their respective peak efficiency points. At 12,000 RPM, R1234yf and R1234ze(E) achieve peak isentropic efficiencies of 0.926 and 0.925 at shaft power requirements of approximately 73 kW, compared to R134a, which achieves an equivalent peak efficiency of 0.926 at 14,000 RPM but requires 106 kW, representing a 31% higher power demand for the same aerodynamic efficiency. The near-identical performance of R1234yf and R1234ze(E) across all metrics is attributable to their almost identical thermophysical properties, particularly molecular weight and specific heat ratio. At 20,000 RPM the pressure ratio difference between the two fluids is less than 0.1 and the isentropic efficiency difference is less than 0.003, confirming that thermophysical properties are the primary governing factor in compressor performance rather than any fluid-specific behaviour. These findings directly support the conclusions of Hmood et al. and Navarro-Esbrí et al. [11,19], who identified R1234yf and R1234ze(E) as the most suitable low-GWP alternatives to R134a, extending their system-level findings to the compressor aerodynamics specifically and providing quantitative evidence that the EU regulatory transition away from R134a and R22 is aerodynamically justified within centrifugal compression systems.

5. Conclusions

Centrifugal compressor performance was assessed using eight working fluids that span the property range relevant to current drop-in substitutions: air, nitrogen, argon, carbon dioxide, R22, R134a, R1234yf, and R1234ze(E). The performance of the compressors was considered through the pressure ratio achieved, the isentropic efficiency of the compressor and the required shaft power. Simulations were performed at mass flow rates between 0.5 and 3.5 m/s and rotational speeds between 10,000 and 20,000 RPM.

A key difference was observed between the operating range of the refrigerants and the non-refrigerants. In particular, the pressure ratios for the refrigerants, for a given rotational speed, are fairly constant over the range of mass flow rates, indicating a broad peak in performance, while the non-refrigerants have a more defined peak in the pressure ratio close to or below the lowest mass flow rate considered. Similarly, the isentropic efficiency of the non-refrigerants has a more pronounced peak at a mass flow rate within the range considered which increases with the rotational speed. In contrast, the refrigerants show a gradual increase in isentropic efficiency with mass flow rate, indicating a peak at a higher mass flow rate. A similar trend is observed in the shaft power, with the peak for the non-refrigerant occurring at mass flow rates within the range which increase with the rotational speed, while the peak for the refrigerants occurs at higher mass flow rates.

Pressure ratio, isentropic efficiency and shaft power all seem to generally increase with the molecular weight of the fluid, the correlation being particularly strong for the pressure ratio. This meant that all the refrigerants produced a higher pressure ratio and isentropic efficiency compared to the non-refrigerants but required a higher shaft power to achieve this. The differences between the lowest and highest values ranged by a factor of 6 for the pressure, and around a factor of 3 for the shaft power. In contrast, the range of efficiency values was only around 3%.

In terms of drop-in use, non-refrigerant gases (air, nitrogen, argon, CO₂) achieved similar peak efficiencies (~0.88) and compatible pressure ratios, indicating they can be used as drop-in substitutes without performance loss; CO₂ performs slightly better due to its properties. For refrigerants, R1234yf and R1234ze(E) match R134a in efficiency while offering higher pressure ratios and significantly lower power requirements at peak efficiency. Their nearly identical performance confirms that thermophysical properties govern compressor behaviour, supporting their suitability as low-GWP replacements.

Author Contributions: Conceptualisation, J.D., J.R.B., J.R. and J.M.B.; methodology, J.D. and J.R.B.; software, J.D. and J.R.B.; formal analysis, J.D., J.R.B., J.R. and J.M.B.; investigation, J.D. and J.R.B.; data curation, J.D. and J.R.B.; writing—original draft preparation, J.D., J.R.B., J.R. and J.M.B.; writing—review and editing, J.M.B. and J.R.; visualisation, J.D.; supervision, J.M.B. and J.R.; project administration, J.M.B. and J.R. All authors have read and agreed to the published version of the manuscript.

Funding: This research received no external funding.

Data Availability Statement: Data supporting the reported results are incorporated into the manuscript.

Conflicts of Interest: The authors declare no conflicts of interest.

References

- Obika, E.; Heberle, F.; Brüggemann, D. From organic rankine cycles to high-temperature heat pumps: A thermo-economic investigation of potential working fluids. *Energy Convers. Manag.* **2025**, *27*, 101156. [CrossRef]
- Gudjonsdottir, V.; Ferreira, C.A.I.; Goethals, A. Wet compression model for entropy production minimization. *Appl. Therm. Eng.* **2019**, *149*, 439–447. [CrossRef]
- Valdés, M.; Sebastián, A.; Abbas, R. Reynolds-number-dependent efficiency characterization of a micro-scale centrifugal compressor using non-conventional working fluids. *Energy Convers. Manag.* **2018**, *177*, 224–232. [CrossRef]
- Xu, P.; Zou, Z. A study of real-gas effect on SCO₂ compressor performance using similitude method. *J. Glob. Power Propuls. Soc.* **2022**, *6*, 200–212. [CrossRef]
- European Parliament and Council of the European Union. Regulation (EU) 2024/573 on fluorinated greenhouse gases, amending Directive (EU) 2019/1937 and repealing Regulation (EU) No 517/2014. *Off. J. Eur. Union* **2024**. Available online: <https://eur-lex.europa.eu/eli/reg/2024/573/oj> (accessed on 1 July 2026).
- United Nations Environment Programme (UNEP). Kigali Amendment to the Montreal Protocol on Substances that Deplete the Ozone Layer. 2016. Available online: <https://ozone.unep.org/treaties/montreal-protocol/amendments/kigali-amendment-2016-amendment-montreal-protocol-agreed> (accessed on 1 July 2026).
- Karber, K.M.; Abdelaziz, O.; Vineyard, E.A. Experimental Performance of R-1234yf and R-1234ze as Drop-in Replacements for R-134a in Domestic Refrigerators. In Proceedings of the Location of International Refrigeration and Air Conditioning Conference at Purdue, West Lafayette, IN, USA, 16–19 July 2012.
- Gokulnath, K.; Krishnan, B.P.; Vijayan, R. Performance Analysis of Alternative Refrigerant Mixtures in Vapour Compression Refrigeration System. *Adv. Nat. Appl. Sci.* **2017**, *11*, 468–475. Available online: <http://www.aensiweb.com/ANAS2017Special11> (accessed on 1 July 2026).
- Sánchez, D.; Andreu-Nácher, A.; Calleja-Anta, D.; Llopis, R.; Cabello, R. Energy impact evaluation of different low-GWP alternatives to replace R134a in a beverage cooler. Experimental analysis and optimization for the pure refrigerants R152a, R1234yf, R290, R1270, R600 and R744. *Energy Convers. Manag.* **2022**, *256*, 115388. [CrossRef]
- Hmood, K.S.; Apostol, V.; Pop, H.; Badescu, V.; Pop, E. Drop-in and retrofit refrigerants as replacement possibilities of R134a in domestic/commercial refrigeration and automobile air conditioner applications. *J. Therm. Eng.* **2021**, *7*, 1815–1835. [CrossRef]
- Belman-Flores, J.M.; Heredia-Aricapa, Y.; García-Pabón, J.J.; Pérez-García, V.; Pérez-Reguera, C.G. Drop-In Replacement of R134a in a Household Refrigerator with Low-GWP Refrigerants R513A, R516A, and R1234ze(E). *Energies* **2023**, *16*, 3422. [CrossRef]
- Yataganbaba, A.; Kilicarslan, A.; Kurtbaş, I. Exergy analysis of R1234yf and R1234ze as R134a replacements in a two evaporator vapour compression refrigeration system. *Int. J. Refrig.* **2015**, *60*, 26–37. [CrossRef]
- Navarro-Esbrí, J.; Mendoza-Miranda, J.M.; Mota-Babiloni, A.; Barragán-Cervera, A.; Belman-Flores, J.M. Experimental analysis of R1234yf as a drop-in replacement for R134a in a vapor compression system. *Int. J. Refrig.* **2013**, *36*, 870–880. [CrossRef]
- Mota-Babiloni, A.; Belman-Flores, J.M.; Makhnatch, P.; Navarro-Esbrí, J.; Barroso-Maldonado, J.M. Experimental exergy analysis of R513A to replace R134a in a small capacity refrigeration system. *Energy* **2018**, *162*, 99–110. [CrossRef]

15. Jawad, M.; Nawaz, I.; Rahil, S.S. Analysis of Low GWP Refrigerants Using VCRC Under Different Conditions for Reduction in Global Warming. Available online: https://www.researchgate.net/profile/Syed-Rahil/publication/376184081_Analysis_of_Low_GWP_Refrigerants_using_VCRC_under_different_conditions_for_Reduction_in_Global_Warming/links/656c8503ce88b87031303f05/Analysis-of-Low-GWP-Refrigerants-using-VCRC-under-different-conditions-for-Reduction-in-Global-Warming.pdf (accessed on 1 July 2026).
16. Kizilboga, K.; Sahin, A.Z.; Zafer, B.; Sahin, I.H. Evaluation of low-GWP refrigerants and their mixtures as potential substitutes for R-134a in terms of safety, performance, and environmental considerations: A review. *SSRN* **2025**. Available online: https://papers.ssrn.com/sol3/papers.cfm?abstract_id=5275842 (accessed on 1 July 2026).
17. Tian, Z.; Wang, C.; Zheng, Q. Investigation of the effects of different working fluids on compressor cascade performance. *Appl. Sci.* **2021**, *11*, 1989. [\[CrossRef\]](#)
18. Xu, C.; Amano, R.S. Empirical design considerations for industrial centrifugal compressors. *Int. J. Rotating Mach.* **2012**, *2012*, 184061. [\[CrossRef\]](#)
19. Meroni, A.; Zühlsdorf, B.; Elmegaard, B.; Haglind, F. Design of centrifugal compressors for heat pump systems. *Appl. Energy* **2018**, *232*, 139–156. [\[CrossRef\]](#)
20. Wang, Y.; Yan, J. Study of Performance Changes in Centrifugal Compressors Working in Different Refrigerants. *Energies* **2024**, *17*, 2784. [\[CrossRef\]](#)
21. Châtel, A.; Verstraete, T. Multidisciplinary optimization of the SRV2-O radial compressor using an adjoint-based approach. *Struct. Multidiscip. Optim.* **2023**, *66*, 112. [\[CrossRef\]](#)
22. Lemmon, E.W.; Ian, H.B.; Huber, M.L.; McLinden, M.O. *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP*; Version 10.0; National Institute of Standards and Technology: Gaithersburg, MD, USA, 2018. [\[CrossRef\]](#)
23. Dickenson, J.; Buick, J.M.; Radulovic, J.; Bull, J. Challenges in CFD model validation: A case study approach using ANSYS CFX and TurboGrid. *Machines* **2025**, *13*, 593. [\[CrossRef\]](#)
24. Eisenlohr, G.; Krain, H.; Richter, F.-A.; Tiede, V. Investigations of the Flow Through a High Pressure Ratio Centrifugal Impeller. In *Proceedings of the ASME Turbo Expo 2002: Power for Land, Sea, and Air, Amsterdam, The Netherlands, 3–6 June 2002*; International Gas Turbine Institute: Houston, TX, USA, 2002; Volume 5: Turbo Expo 2002, Parts A and B, pp. 649–657. [\[CrossRef\]](#)

Disclaimer/Publisher’s Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.